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POTENTIAL OF DIESEL ENGINE, EMISSION TECHNOLOGY

Joseph Sturm
Thomas Trella

U.S. DEPARTMENT OF TRANSPORTATION
Research and Special Programs Administration
Transportation Systems Center
Cambridge MA 02142

MARCH 1980
FINAL REPORT

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This report surveys diesel engine emission technologies applicable to passenger cars and light trucks. The general design and operating features are presented and discussed. Current and state-of-the-art concepts are reviewed with the focus on control of diesel emissions through (1) modification of the combustion process, (2) aftertreatment systems and (3) fuel modifications.

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PREFACE

This report, DOT-TSC-NHTSA-79-40, is one of a series of four companion reports to DOT-TSC-NHTSA-79-38 "Potential of Diesel Engine, 1979 Summary Source Document."* It presents state-of-the-art concepts on control of diesel emissions. This report is a deliverable under PPA HS-027 "Support for Research and Analysis in Auto Fuel Economy and Related Areas."

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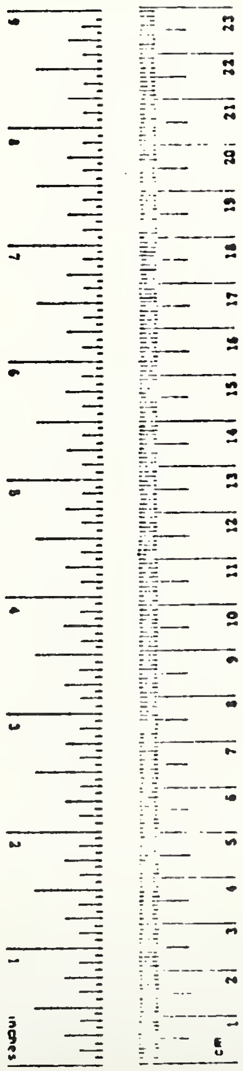
*"Potential of Diesel Engine, 1979 Summary Source Document," by T. Trella, Report No. DOT-TSC-NHTSA-79-38, March, 1980 .

METRIC CONVERSION FACTORS

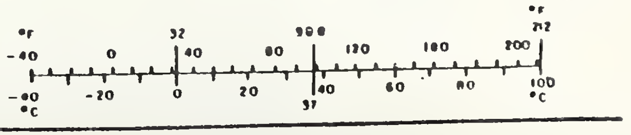
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.

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INTRODUCTION

The control of the undesirable diesel engine pollutants might limit the potential improvement in fuel economy expected from the increased penetration of the diesel engine in the transportation field. These pollutants include the regulated emissions of nitrogen oxides, carbon monoxide and hydrocarbons and the nonregulated emissions such as the particulate, polynuclear aromatic, odor, and irritant producing compounds.

These pollutants are the result of the autoignition and combustion processes and the complexities which result from the use of heterogeneous mixtures in diesel engines. Unfortunately, the mechanisms of autoignition, combustion, and emission formations in diesel engines are not well understood because of the complexities of the instrumentation and the length of time needed for such investigations.

This document surveys diesel engine emissions technologies applicable to passenger cars and light trucks. The general design and operating features are presented and discussed. Current and state-of-the-art concepts are reviewed with the focus on control of emissions through (1) modification of the combustion process, (2) aftertreatment systems and (3) fuel modifications.

1. ENGINE CHARACTERIZATION

Categorization of reciprocating internal combustion engines is generally subdivided into homogeneous combustion and heterogeneous combustion systems. Conventional gasoline spark ignition engines employ homogeneous combustion systems, whereas, the diesel and gasoline stratified charge engines are of the heterogeneous type. The homogeneous combustion system relies on assisted ignition (spark plug) of a well prepared fuel-air mixture from which its name is derived. Fuel is provided by means of a carburetor or injection system and is mixed with air in the engine's induction system before entering the combustion chamber. The fuel-air charge lies between the lean and rich flammability limits, and combustion takes place in the form of a flame front starting at the spark plug and propagating through the homogeneous charge.

In a heterogeneous combustion system, the fuel-air mixture is not prepared in the induction system, but fuel is delivered or injected directly into the combustion chamber where it is simultaneously mixed with the air.

In such engines as in the stratified charge and diesel heterogeneous combustion, the overall fuel-air charge is generally fuel-lean. The fuel/air mixture ratio, however, may vary from one part of the charge to another. In the stratified charge engine ignition starts by an electric spark located in a rich fuel zone. The rest of the combustion process depends upon the design of the combustion chamber and whether it is one chamber or a divided chamber. In the divided chamber, the spark occurs in the prechamber where the mixture is rich and a torch from the prechamber ignites the charge in the main chamber, which is lean. The diesel engine can be considered a stratified charge engine which uses heat of compression for ignition. Such engines display high thermal efficiency mainly caused by the relatively high compression ratio required to start the autoignition process, the lower pumping losses, as a result of the absence of the throttle valve (which is needed in spark ignition enginesto control the power output) and the overall lean mixture required to achieve a satisfactory heterogeneous combustion process.

2. DIESEL ENGINE TECHNOLOGIES (GENERAL)

Diesel combustion can take place in one chamber (direct injection or open chamber) or in a divided chamber (indirect injection). The General Motors Oldsmobile 350 diesel and Peugeot 504D are divided chamber types; the Mercedes Benz 240D and 300D are divided chamber types with prechamber design.

2.1 DIRECT INJECTION ENGINES

2.1.1 Mixture Formation - In D.I. or open-chamber (O.C.) diesel engines, the fuel is injected directly into the bulk of air in the combustion chamber. The mixing of fuel and air is attained by using the kinetic energy of the fuel jet flowing out of the nozzle orifices, and the energy of the moving air charge. The major types of air motion are the rotary (swirl), and the radially inward (squish) components.

The swirl air motion is generated when the cylinder is being charged with the fresh charge as the air flows through directed intake ports, and sometimes through masked intake valves or ports, in a tangential direction to the cylinder. The resulting rotary motion or swirl continues during the compression stroke in the form of a forced vortex solid-body rotation[1], and is augmented by the flow of the charge into the combustion space, which has a smaller diameter than the cylinder, as the piston approaches the cylinder head. The radial velocity component is generated by the inward flow of the charge into the combustion chamber as the piston approaches TDC. The combustion chamber geometry, such as bowl diameter, bowl throat diameter, flank angle, lip shape and bowl shape have been found to have a great effect on the rate of fuel and air mixing, and thus the performance and emission characteristics of the engine.[2]

In some designs, helical intake ports are used to produce a rotation of the air around the stem of the valve before entering the cylinder. List[3] introduced a fin mounted in the intake port to divide the airflow into at least two subsidiary currents flowing in opposite directions, to increase turbulence in the cylinders of D.I. engines.

Excessive fuel atomization results in poor penetration and in inefficient air utilization and combustion. Excessive air swirl also results in less penetration and overlap of the sprays which results in poor combustion. Therefore, the optimum swirl may vary from one engine to another depending upon the design of the combustion chamber, the number and size of injection holes, and other characteristics of the injection system.[4,5,6] The fuel injection pressures in D.I. engines vary from about 3000 psia to 8500 psia. These pressures usually exceed the critical pressures of the majority of the fuel constituents (311 psia for n-decane and 272 for n-dodecane). Actual diesel fuels contain heavier hydrocarbons than those mentioned above. Their critical pressures decrease with the increase in the number of carbon atoms in the molecule. Upon injection, the fuel pressure drops to the cylinder air pressure which is in the range of 350-850 psia.[7] Higher pressures may be reached in supercharged engines.

In many engines, particularly those with turbocharging, fuel evaporation in the cylinder may occur under supercritical conditions. Most of the previous studies on droplet evaporation have been concerned with subcritical evaporation which occurs in low compression ratio, naturally-aspirated diesel engines. Studies in supercritical evaporation and combustion have been limited to pure liquid compounds.[8,9,10,11] In diesel engines, the droplets consist of a very large number of compounds with quite different thermodynamic properties. One major problem in the analysis of actual petroleum fuels is that of finding an average property for distillates. One of the methods proposed for such studies is that of the "Characterization Factor",[12] however, there is no way to check the validity of using this method for the study of supercritical evaporation of actual fuels in engines.

A limited number of studies have been made on the droplet size distribution in fuel sprays in D.I. diesel engines. Basic studies are needed in this area to utilize modern electronic and holographic techniques.

In multicomponent liquid sprays, the concentration of the higher-boiling components in the liquid drops increases as the spray evaporates.[13] At higher air temperatures, this trend toward increased concentration of the higher-boiling components is more pronounced.[13] This means that the lower boiling point components are the first to evaporate and mix with the air, followed by the heavier components. Therefore, in diesel engines, the mixtures are not only heterogeneous, but also have different fuel compositions in the different parts of the spray, at different intervals during the combustion process.

2.1.2 Autoignition - Unlike the conventional and the stratified charge gasoline engines, in which combustion starts by an electric spark at one location, the combustion in the diesel engine starts by autoignition nuclei at numerous locations in the combustion chamber.

The preignition processes in diesel engines may be divided into physical and chemical processes. The physical processes are:

(1) spray disintegration and droplet formation,

(2) heating of the liquid fuel and evaporation, and

(3) diffusion of the vapor into the air to form a combustible mixture.

The chemical processes are:

(1) the decomposition of the heavy hydrocarbons into lighter components, and

(2) the preignition chemical reactions between the decomposed components and oxygen.

The time taken after the start of injection for the preignition processes to produce the ignition nuclei and detectable combustion phenomena, is known as the ignition delay. The duration of the ignition delay is one of the most important criteria, having a great effect on the combustion process, mechanical stresses, engine noise, exhaust gaseous and particulate emissions.

In high speed automotive diesel engines, the preignition chemical processes have been found to be the rate-controlling processes during the ignition delay.[14] However, very little is known about their detailed mechanisms. More research work is needed, particularly in the area of mechanisms which lead to the formation of aldehydes, oxygenated hydrocarbons, odor forming compounds, and particulates.

2.1.3 Combustion - The combustion process in D.I. engines may be considered to take place in two stages. The first stage follows the formation of the ignition nuclei, and results in the combustion of the already-mixed fuel vapor and air during the ignition delay period. This part has been found to have a major contribution in the NO_x formation.[15] The second stage takes place in the diffusion type flame combustion in the core of the spray and the fuel deposited on the walls. This stage results in the formation of the incomplete combustion products such as the unburned hydrocarbons, partially oxygenated hydrocarbons, and particulate emissions.

2.2 INDIRECT INJECTION ENGINES

In this type of engine, fuel is injected into the prechamber. Mixing is attained mainly by using the kinetic energies produced by: (a) the flow of air into the prechamber during the compression stroke, and (b) the flow of the products of combustion from the prechamber to the main chamber. The kinetic energy of the fuel jet flowing out of the nozzle orifices contributes to the mixture formation to a much smaller degree than in the D.I. engines. Accordingly, the injection pressures in the divided chamber engines can be lower than those in the open chamber engines. Other advantages of the I.D.I. design, resulting from the higher energy of mixing, are better air utilization (A/F equivalence ratio = 1.15 to 1.25) and higher specific output and rated speeds. Furthermore, the maximum pressures, rates of pressure rise, and noise level in the I.D.I. engines are lower than in the D.I. engines. Because of the high turbulence and surface to volume ratio, however, the cooling losses are higher than in the D.I. engine. Other disadvantages of the I.D.I. engine are the lower thermal efficiency, and the higher local thermal stresses caused by the impingement of the high velocity gases discharged from the prechamber into the main chamber. I.D.I. engines can use a single orifice spray, are suitable for small size engines (as small as 250 c.c./cylinder), and can run at up to speeds of 5000 r.p.m.[16]

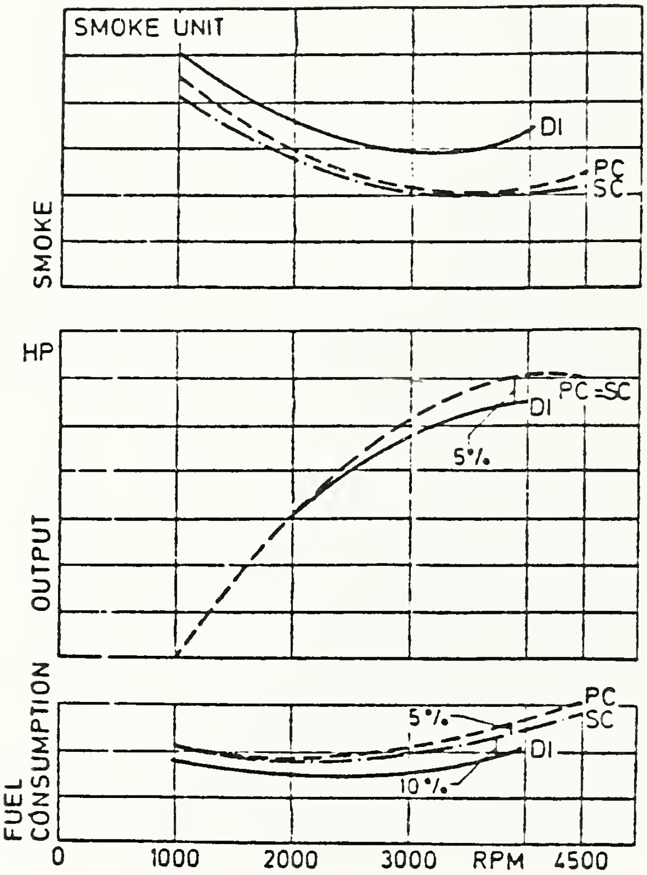
The performance of the I.D.I. engine depends upon many factors, including the ratio of the prechamber volume to the total clearance volume, the eddies and gas flow patterns in the prechamber and main chamber, size and direction of the throat, piston recesses, and the surface temperatures.

The "swirl" type of I.D.I. engine is widely used in light-duty diesel engines with cylinders under 900 c.c. In the "swirl" type, the volume of the prechamber is about 50 percent or more of the total clearance volume. A well organized, directed swirl motion is produced in the prechamber during the compression stroke by the flow of air through a tangential throat. The swirl increases during the compression stroke, and reaches its maximum a few degrees before Top Dead Center (T.D.C.). The swirl measured by Lyn[17] in a Comet Mark V, reached its maximum at 7 degrees before T.D.C. by using the Schlieren techniques, and was about twenty-one times the engine speed. The swirl helps to mix the fuel and air before the start of combustion in the prechamber. After ignition, the flame moves toward the center of the prechamber due to the increase in the buoyant force acting on the hot products of combustion in the centrifuga field created by the swirl.[18] As the combustion process proceeds, and the pressure increases in the swirl chamber above that in the main chamber, the flow and the swirl directions are reversed. The resulting turbulence in the swirl chamber and flow into the main chamber help mixing and combustion. Nagao[19] and Walder[20] found that the direction of injection with respect to the throat has a great effect on engine performance and smoke emissions. Bowdon[21] found that in swirl chambers the ratio of the heat released in the prechamber to the total heat released was the same as the ratio of the prechamber volume to the total clearance volume. Other performance characteristics of the swirl chamber may be found in reference 22.

In other types of I.D.I. engines, the volume of the prechamber and the area of the throat are smaller than in the swirl type. In these cases, the air motion is not tangential to the prechamber, and the fuel is injected along its axis toward the connecting passage. The main turbulence is produced after the start of combustion by the flow of the products from the prechamber into the main chamber where the combustion is completed. The number and shape of the connecting passages affect the extent of mixing of the products from the prechamber and the air in the main chamber.

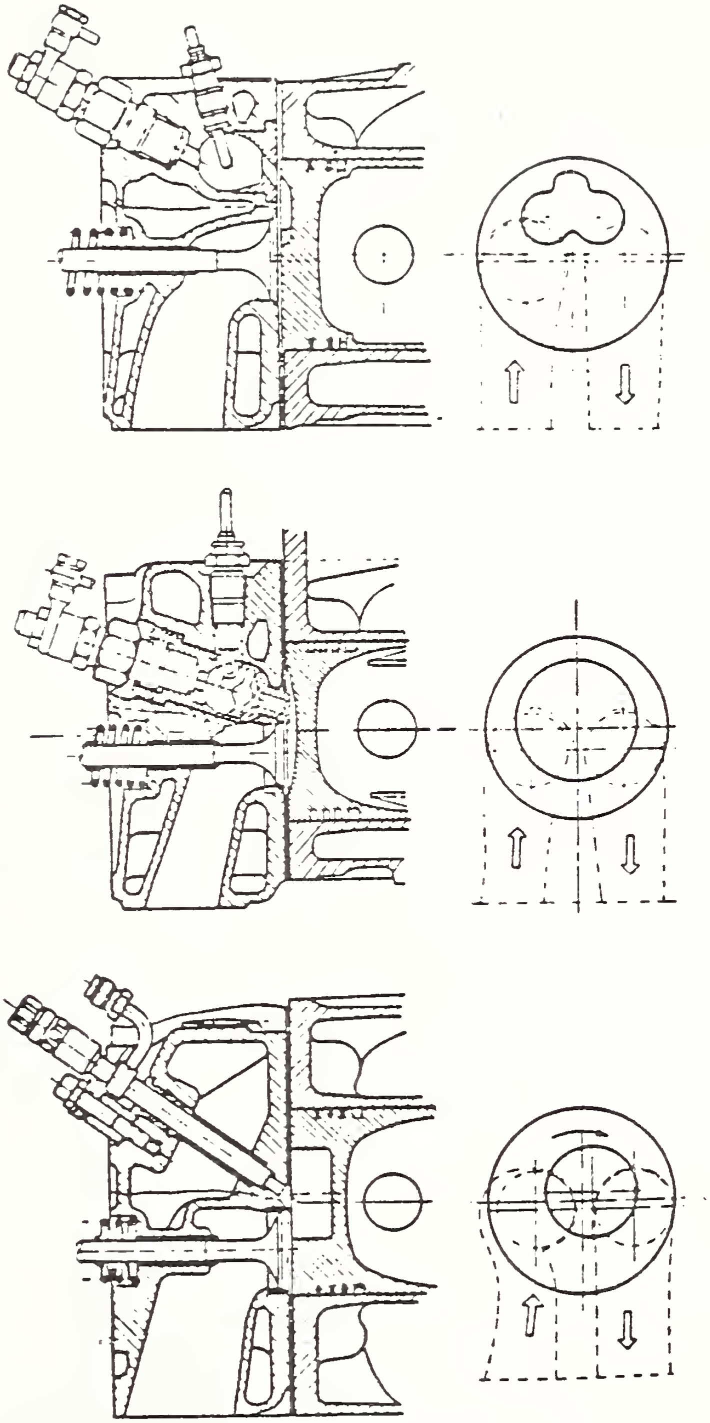
A comparison between the performance and smoke of three different combustion chambers for light-duty diesel engines (Daimler Benz) is given in Figure 1, taken from Reference 23. The configuration of the three combustion chambers is given in Figure 2. Figure 1 shows that the D.I. engine has the best fuel economy but the highest smoke. The swirl chamber has about a 5 percent advantage in fuel economy over the P.C. engine because of the lower pumping losses between the swirl chamber and the main chamber.

In a recent study by Middlemis,[2] comparisons were made between an I.D.I. prechamber, a D.I. open chamber and a D.I. re-entrant chamber. This study showed that the new re-entrant chamber maximum thermal efficiency occurs at a more retarded injection timing than the open chambers. This contributes to the lowering of the NO_x and smoke emissions.



Source: Reference 23.

FIGURE 1. COMPARISON OF OUTPUT, FUEL CONSUMPTION AND SMOKE OF LIGHT DUTY DIRECT INJECTION (D.I.), PRE-CHAMBER (P.C.) AND SWIRL CHAMBER (S.C.)



SWIRL CHAMBER

PRECHAMBER

DIRECT INJECTION

Source: Reference 23.

FIGURE 2. DIFFERENT COMBUSTION SYSTEMS FOR LIGHT-DUTY DIESEL ENGINES (DIAMLER-BENZ)

3. DIESEL ENGINE OPERATING CHARACTERISTICS

3.1 EMISSIONS

The concentration of the different emission species in the exhaust is the result of their formation, and their reduction or further formation in the cylinder and exhaust system. The incomplete combustion products formed in the early stages of combustion may be oxidized later during the rest of the combustion process if they mix with oxidizing gases at a high enough temperature and for the proper residence time. In most cases, the NO formed is not decomposed, particularly in the lean combustion zones, but NO may increase in concentration during the rest of the combustion process if the temperature remains high or becomes higher. If no further oxidation takes place, some decomposition may occur in the NO formed in the fuel-rich regions.

3.1.1 Unburned Hydrocarbons

The unburned hydrocarbons in the diesel exhaust consist of either original or decomposed fuel molecules or recombined intermediate compounds. A portion of these hydrocarbons originates from the lubricating oil.

The hydrocarbon emissions in I.D.I. engines are lower than in D.I. engines because of the more effective mixing which results from the flow of the combustion products from the prechamber to the main chamber.

3.1.2 Carbon Monoxide

Carbon monoxide is one of the compounds formed during the intermediate combustion stages of hydrocarbon fuels.[24] As combustion proceeds to completion, oxidation of CO to CO_2 occurs through recombination reactions between CO and the different oxidants. If these recombination reactions are incomplete due to lack of oxidants or due to low gas temperatures, CO will be left without oxidation.

Carbon monoxide emissions in I.D.I. are less than in D.I.

3.1.3 Smoke and Other Particulates

3.1.3.1 Types - Different types of particulates are emitted from diesel engines under different modes and operating conditions. These particulates can be divided into the following:

(a) Liquid particulates appear as white/blue clouds of vapor emitted under cold starting, idling, and low loads. These consist mainly of fuel and a small portion of lubricating oil[25] emitted without combustion; they may be accompanied by partial oxidation products. The white/blue clouds disappear as the load is increased and the cylinder walls become warmer.

(b) Soot or black smoke is emitted as a product of the incomplete combustion process, particularly at high load conditions. The black smoke emissions consist of irregularly-shaped agglomerated fine carbon particules.

(c) Other particulates include lubricating oil and fuel additives.

3.1.3.2 Physical Properties - The particles have sizes ranging from 0.04 µm to 30 µm[27]. The smallest particles are spherical and form aggregates to give the wide distribution in the exhaust. The irregularly shaped agglomerated particles appear as "fluffy heaps" which are difficult to disperse, have a bulk density of 0.075 gm/cm^3, and have a large surface-to-volume ratio in the range of 25-50 m^2/m^3. Hydrocarbons can be absorbed on the surface.

3.1.3.3 Chemical Properties - The particulates consist of carbon, high-molecular-weight unpyrolized paraffins, oxy-, hydroxy-, and carboxy paraffins, and nitrite and sulfate salts. The carbon is a family of complex polybenzenoid compounds, always containing hydrogen, or 1-3 percent by mass.[28]

3.1.4 Nitrogen Oxides NO_x

Most of the nitrogen oxides in the exhaust are in the form of nitric oxide. Approximately 10 percent of the NO_x appear as NO_2. Many mechanisms for NO formation in combustion systems have been proposed.[32 to 36] The widely accepted mechanism is that of Zeldovich[32] shown in Equations (2) and (3).

$$O_2 \rightleftarrows 2O \qquad \text{(Eq. 1.)}$$

$$O + N_2 \rightleftarrows NO + N \qquad \text{(Eq. 2)}$$

$$N + O_2 \rightleftarrows NO + O \qquad \text{(Eq. 3)}$$

$$N + OH \rightleftarrows NO + H \qquad \text{(Eq. 4)}$$

The chain reactions are initiated in Equation (2) by the atomic oxygen which is formed from the dissociation of oxygen molecules at the high temperatures reached in the combustion process. According to this mechanism, the nitrogen atoms do not start the chain reaction, because their equilibrium concentration during the combustion process is relatively low compared to the equilibrium concentration of atomic oxygen. Therefore, in diesel combustion, the local NO formation in the spray is related to the local oxygen atom concentration. This is a function of the local concentration of the oxygen molecules and the local temperature.

In the diesel engine, NO is not formed during the compression stroke, even in very highly supercharged engines, because of the relatively low temperatures reached.

3.1.5 Noise

One of the objectionable pollutants from many combustion engines, particularly diesel engines, is noise. Diesel engine noise, therefore, has been subject to many investigations during the last few years.[16,22,37-47]

The major exciting forces, which set the engine structure into vibration and result in noise, are mainly produced from the combustion process. Other sources of noise include the fuel injection system, air intake and exhaust, timing gear, and auxiliaries such as cooling fan, supercharger or turbocharger, etc.

Priede[48] showed that the exciting forces produced from the cyclic cylinder pressure can be expressed in terms of a frequency spectrum of the pressure diagram by using a Fourier series. Priede found that the spectrum of the diesel engine may be considered to be composed of three major parts.[47] The first is a broad peak in the low frequency range centered around the fundamental repetition frequency of combustion. Its magnitude is mainly a function of the peak gas pressure and the width of the pressure diagram (at 2/3 peak pressure). The gasoline engine has the lowest peak pressure during idling and shows the lowest level. The second part is a linearly decaying part in the acoustically important region around 1000 Hz (400-3000 Hz). The typical slope of this line for naturally aspirated diesel engines is 30 dB/decade. That is, an increase in engine noise of 30 dB is expected from a tenfold increase in engine speed. (The corresponding increase for S.I. engines is 45 dB/decade.) The slope of this line is a function of the rate of pressure rise after the end of the delay period. The diesel engine has higher levels than the gasoline engine at frequencies above 800 Hz. These higher levels in the diesel engine are attributed to the high rate of pressure rise at the end of I.D. as compared to the smooth pressure rise in the gasoline engine caused by the propagation of the flame through the premixed combustible charge.

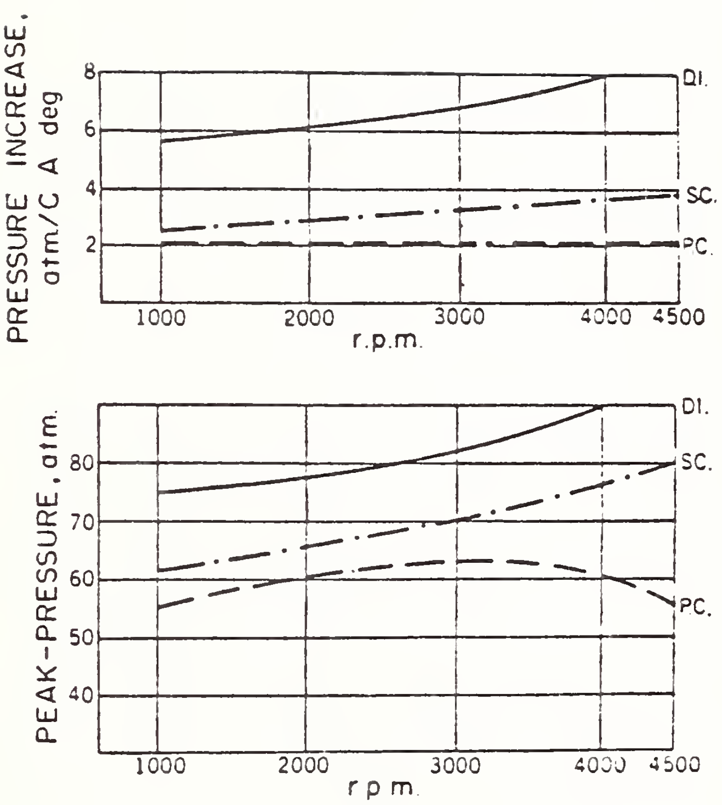
A reduction in combustion-generated noise in D.I. engines in the critical midfrequency range (400-3000 Hz) can be achieved by shortening the ignition delay period, which results in a lower rate of pressure rise and maximum pressure, and in a smoother pressure trace. Among the methods known to shorten the delay period and reduce the engine noise are: injection retard, injection rate modifications, turbocharging, increased compression ratio, the use of pilot injection, and the use of a higher cetane number fuel. The third part of the spectrum has a peak around the 5000 or 6000 Hz frequency range. This excitation peak is caused by wave motion in the combustion chamber, which appears as fluctuations in the pressure trace at the peak gas pressures. Tiede et al.[41] found that these excitation levels take place in the 5000-10000 Hz frequency range and that they vary greatly from cycle to cycle. Tiede concluded that at current engine noise levels they do not contribute significantly to overall noise levels.

Austen and Priede[16] found that the different noise characteristics of diesel and gasoline engines are due to differences in excitation by different forms of cylinder pressure, and not due to structural differences. In diesel engines, the sound pressure level at no-load differs only slightly from that at full load because the compression pressure is almost the same. In gasoline engines, where engine throttling is applied, the sound pressure level at no-load is less than that at full load.[16]

Austen and Priede [48,49] indicated that reduction in engine noise can be achieved through combustion modification if the cylinder pressure is above the critical pressure for the engine under consideration. The critical pressure is defined as the pressure at which the noise due to combustion has attained the same level as the general mechanical noise. If the cylinder pressure level is below the critical value, further smoothing of the cylinder pressure has a negligible effect on engine noise. In D.I. engines, noise is higher than in the other types because of its higher maximum cylinder pressures and rates of pressure rise. This is shown in Figure 3 (from Reference 23) for three different light duty diesel engines.

Monoghan, et al.,[22] indicated that the problem of diesel noise could be largely eliminated if the number of cylinders were increased from four to six or eight. Here, the idle noise would be less objectionable, but would still be very pronounced as compared with a gasoline engine.

In addition to reducing noise at the source, noise may be reduced through structural modifications to the crankcase, cylinder block casting, valve gear cover and oil pan[37,38,41,43,50]



Source: Reference 23.

FIGURE 3. COMPARISON BETWEEN THE PEAK PRESSURE AND THE RATE OF INCREASE IN PRESSURE DUE TO COMBUSTION IN THE DIRECT INJECTION (D.I.), SWIRL CHAMBER (S.C.), AND PRE-CHAMBER (P.C.) AUTOMOBILE ENGINES

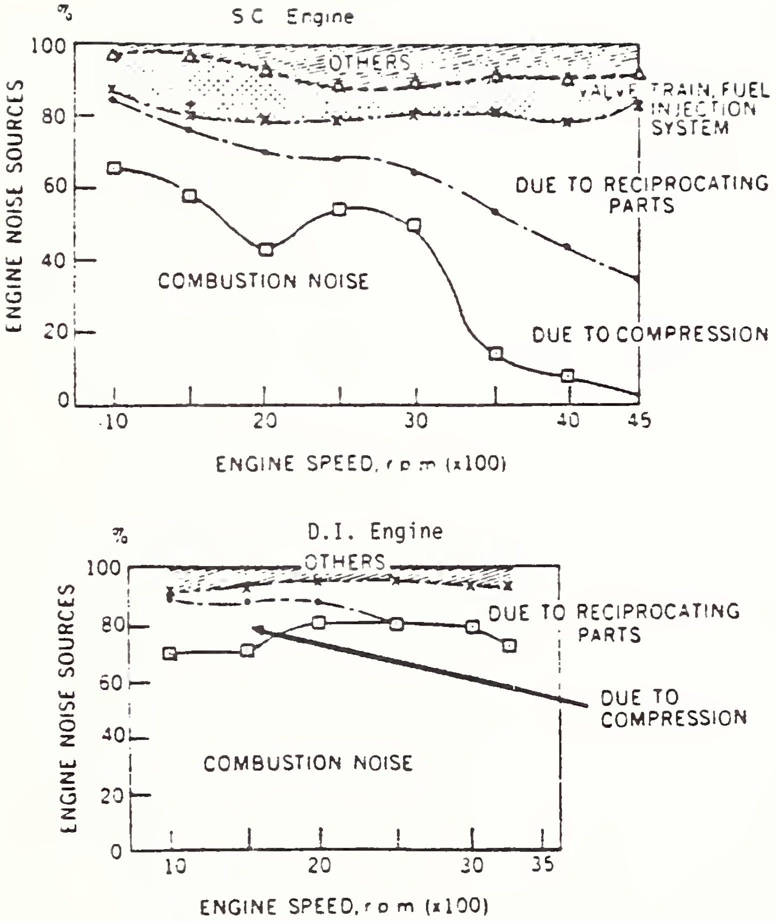
or through enclosing the engine.[39,50]

Kithara et al.[50] made an analysis of sound powers as a percentage of total engine noise for each noise source in an S.C. (swirl chamber) engine and a D.I. engine, as shown in Figure 4. For the S.C. engine, combustion noise accounts for 60-70 percent of the noise at low speeds, and mechanical noise is predominant at high speeds. Meanwhile, for the D.I. engine, combustion accounts for 70-80 percent of the total noise over the entire r.p.m. range. Kithara reported that since mechanical noise in the main source of noise in the S.C. engine, the closed shielding of the cylinder body and oil pan have been very effective in noise reduction. Meanwhile, it was necessary to retard the timing of the D.I. engine in addition to providing oil pan shielding to reduce its noise effectively.

3.1.6 Odor

Odor is one of the diesel engine's nuisance type pollutants which the public finds objectionable. It has been the subject of many investigations during the last few years, particularly as it relates to odor-producing species and instrumentation.[23,51-61] A review of diesel odor research was made by Savery et al.[62]

Diesel exhaust odor can be divided into two characteristic types: oily-kerosene and smoky-burnt. The concentration of the odor producing species in diesel exhaust is very small.[56] Early research was directed toward identifying the individual odorous species. Kendall et al.[59] found that groups of similar components, rather than individual species, contribute to the characteristic diesel exhaust odor. Spindt, et al.[56] identified many mono- and poly-oxygenated partial oxidation products and certain fuel fractions as being the odor-producing compounds. The DOAS[62] (Diesel Odor Analysis System) developed by Arthur D. Little, Inc., is based on the fact that two groups of compounds produce odor. The first group of aromatics such as alkyl benzenes, indenes, indans, tetralins, napthalenes and etc. The second is a group of oxgenates.



Source: Reference 56.

FIGURE 4. NOISE SOURCES IN A SWIRL CHAMBER (S.C.) AND A DIRECT INJECTION (D.I.) DIESEL ENGINE

According to Reference 53, the principal components responsible for the characteristic oil-kerosene portion of diesel exhaust odor are alkyl benezenes, indans, tetralins and indenes. The contribution of the alkyl napthalenes, which constitute a major portion of the mass of the oil-kerosene fraction, to odor perception may be through a synergistic effect. The unburned fuel in the exhaust is very likely to contribute heavily to the oily-kerosene odor.

Somers and Kittredge[55] indicated that alkyl-substituted benezene and napthalene type compounds have been related to the oily-kerosene odor quality, and that oxygenated aromatic structures have been related to the smoky-burnt odor quality. O'Donnell and Dravnieks[54] found from mass spectral data that in addition to the partial oxidation of hydrocarbon compounds, sulfur species are among the important odor contributors. Later work[56] indicates that sulfur compounds in the fuel are not important odor contributors. Padrta and Samson[64] observed that oxides of nitrogen contribute to diesel odor, but were not the true culprits.

Odor intensities are high at no load and full load,[65] are similar for 2-stroke and 4-stroke engines,[51] decrease with lowered aromatic content of the fuel,[22] and can be reduced by use of catalytic reactors.[51,64] Data published by Merrion[66] showed that the odor intensity was not affected by engine speed, increased with increased load, and decreased with improved design of the injection system. Rounds and Pearsall,[65] and Merrion[66] also suggested that there may be some weak correlation between formaldehyde emissions and odor intensity. Some experiments were made in an effort to identify the mechanism by which the odorous compounds are formed in the combustion processes. Barnes[67] found that large differences in exhaust odor intensity were achieved with the same fuel by altering the intake atmosphere conditions on an experimental engine. A single cylinder, 4-stroke cycle diesel engine was employed using n-heptane as a fuel. The artifical atmospheres supplied to the engine were comprised of air or of oxygen plus an inert gas (argon, helium, nitrogen, and carbon dioxide). These mixtures of inert gases and oxygen have different lean ignitability limits. Barnes found that there is a correlation between the odor levels and the lean flammability limits, LFL. The LFL ratio is defined as the ratio of percentage of fuel by volume at the LFL with the mixture to the percentage with air. An LFL ratio more than unity means that the fuel/oxidizer ratio is richer at the LFL than in air. Barnes[67] concluded that for such mixtures, a larger portion of the injected fuel would be too lean to burn and would be partially oxidized. He observed that this was particularly true at idling and light loads.

3.1.7 Startability

I.D.I. diesel engines which are used in passenger cars and other light duty applications are more difficult to start than D.I. engines. However, they lend themselves to the use of glow plugs to heat the air and fuel in the prechamber. Because the glow plug should reach a high temperature before cranking the engine, there is some delay in starting. The development of high intensity glow plugs can help to reduce the starting delay.

3.2 LOW POWER-WEIGHT RATIO

The low power/weight ratio causes the diesel powered car to take a longer time for acceleration. The diesel engine produces less maximum power than a gasoline engine of equal displacement for two reasons. First, the overall fuel-air ratio in the diesel is always leaner than the stoichiometric ratio. The limit to the increase in fuel-air ratio is smoke production. Second, the maximum rated speeds in diesel engines are less than in gasoline engines. The limit here is the short time allowed for fuel injection, evaporation, mixing and combustion for proper engine operation at higher speeds. Mechanical stress considerations also limit the speed of the diesel engine. In addition to the larger displacement for the same power output, the components of the diesel engine are heavier than the corresponding ones in the gasoline engine.

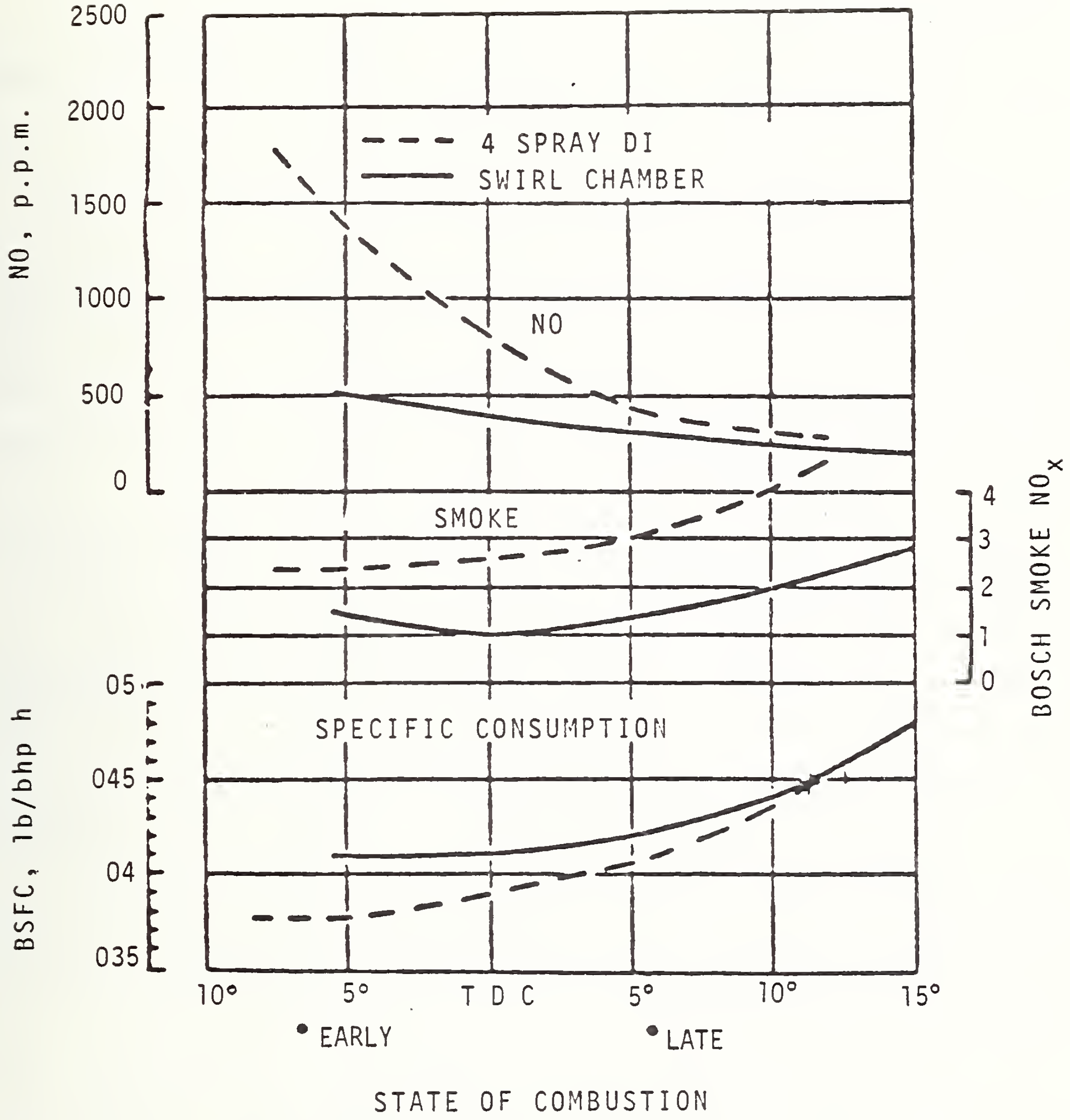
4. EMISSION CONTROL PARAMETERS

Many problems are encountered in controlling the emissions from any combustion engine, because the controls must simultaneously reduce the emission of more than one pollutant into the atmosphere. Unfortunately, many controls which reduce the concentration of the incomplete combustion products such as hydrocarbons, carbon monoxide, and particulates cause an increase in NO_x: NO is a product of an efficient combustion process. The problem is difficult for the diesel engine because many of these controls (particularly for NO_x) reduce the fuel economy, which is a major attraction for the diesel engine. The emission control may be achieved by modifying the combustion process, after treatment or fuel additives.

4.1 COMBUSTION MODIFICATION

4.1.1 Injection Timing

Retarding the injection timing is very effective in reducing the NO_x emissions in both D.I. and I.D.I. engines. Walder[20] compared the effect of injection timing on NO, smoke and fuel economy in a D.I. and in a swirl chamber engine. Figure 5 indicates that, at a sacrifice in fuel economy, the swirl chamber produces less NO and smoke than the D.I. engine. Solid particulate emissions increase with injection retard in the D.I. engine, but they can be reduced by increasing the injection pressures[70] and by setting the timing other than T.D.C. in the swirl chamber engine. Pischinger[71] found that hydrocarbons increased in D.I. and I.D.I. engines when injection was retarded. This effect was more pronounced in D.I. than in I.D.I. engines. In some cases, Springer and Dietzmann[52] found that retarding the injection timing in a D.I. engine had no apparent effect on odor or CO and increased the unburned hydrocarbons. In another D.I. research engine[71], retarding the timing beyond the best economy point increased the CO and smoke. V.W.[72] results show that when the



Source: Reference 20

FIGURE 5. COMPARISON BETWEEN SMOKE NO_x AND FUEL ECONOMY ON DIRECT INJECTION (D.I.) AND INDIRECT INJECTION (I.D.I) SWIRL CHAMBER ENGINES

injection timing is retarded, composite fuel economy is reduced, and the HC and CO composite emissions are increased.

4.1.2 Water Addition

This section discusses the effect of water addition on emissions for direct (D.I.) and indirect (I.D.I.) injection diesels.

4.1.2.1 D.I. Engines - Adding water changes the lean ignition limit and increases the hydrocarbon and CO formation. It also reduces the oxidation reactions due to the drop in the maximum temperatures, which increases in the hydrocarbon and CO emissions. Increasing inlet air humidity in a D.I. engine was found by Springer and Dietzmann[52] to reduce NO significantly and to increase smoke. Its effect on hydrocarbons and CO was minor except at full load.

Valdmanis and Wulfhorst[73] compared the effect of introducing water in the fuel as an emulsion to water introduced with the intake air in a D.I. engine. As the ratio of water to fuel increased, the I.D. increased and injection had to be advanced to obtain peak power. These requirements were greater with the emulsified fuel than with induced water. It seems that the NO emissions in these tests were affected by two factors: the effect of water on reducing the maximum temperatures and oxygen concentration, and the effect of injection advance on the maximum temperatures reached. These two factors resulted in an increase in NO emissions with the emulsified fuel and a decrease in NO emissions with the induced water.

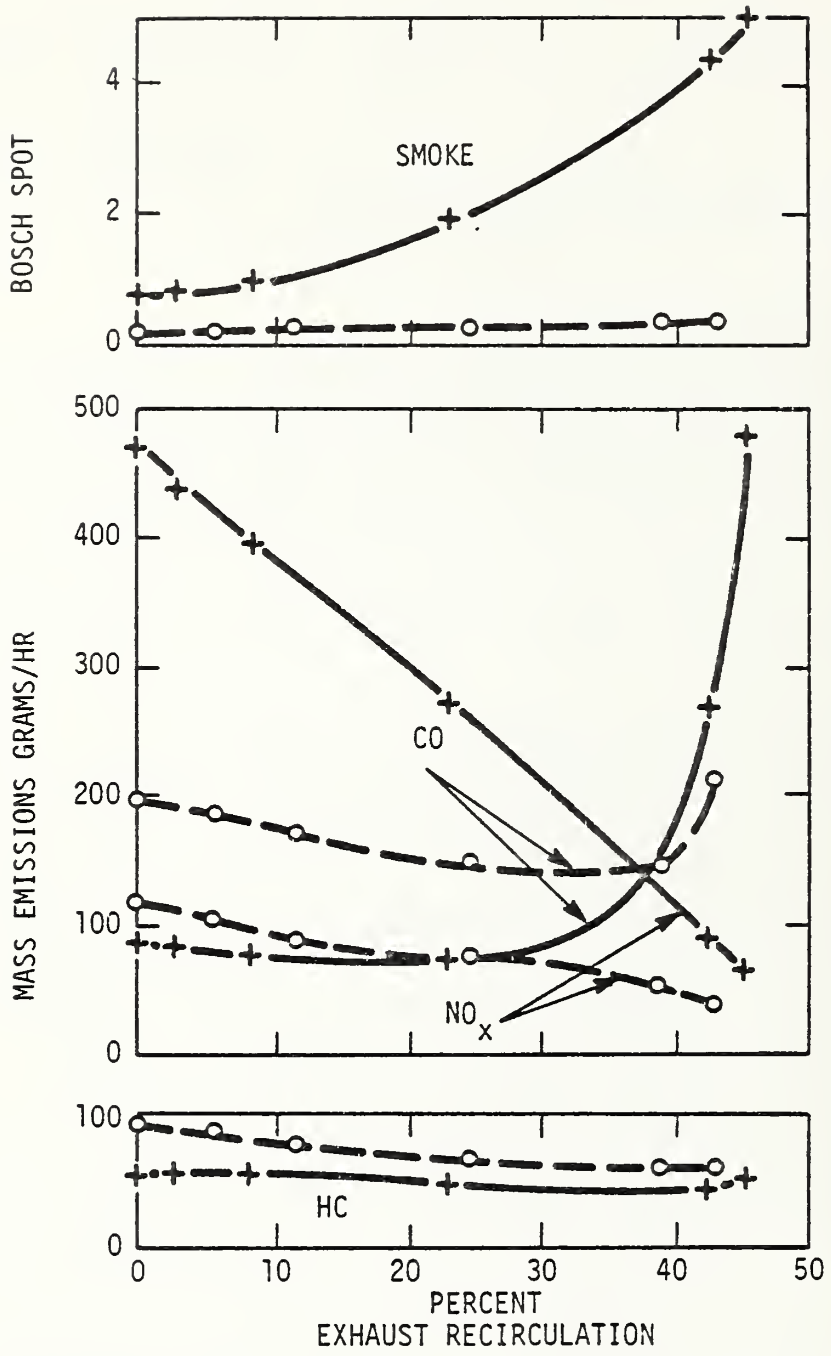
The combined effects of water addition and injection advance (for optimum power) in Valdmanis' tests affected the other emissions in different ways. It resulted in a reduction in smoke and in an increase in unburned hydrocarbon emissions.

4.1.2.2 I.D.I Engines - The effect of inlet manifold water injection on a swirl type I.D.I. engine, Comet V, was studied by Torpey et al.[74] They found that water injection decreased the NO and had virtually no effect on the CO and hydrocarbon emissions, except at full load where they increased.

4.1.3 Exhaust Gas Recirculation

The reduction in NO emissions in diesel engines by exhaust gas recirculation (EGR) is believed to be due to the increase in the heat capacity of the charge, as has been found in gasoline engines.[75,76] EGR affects the combustion and emissions in the D.I. and I.D.I. engines in a manner similar to water injection. The effect of percentage EGR on the NO, smoke, unburned hydrocarbon, and CO emissions and on fuel consumption for a D.I. engine has been studied by Pischinger et al.[71] and is shown in Figure 6. This figure shows that the hydrocarbons and NO_x decrease with increases in EGR. The CO decreases with the increase in EGR to a point after which any increase in EGR results in an increase in CO. Smoke increases slightly with increases in EGR at no load, but at an increasing rate at part load. Similar effects of EGR on reducing the NO emissions have been observed by Walder[20] and by Abthoff and Luther[77] in two types of I.D.I. engines.

Tests on automotive prechamber engines showed that EGR effects NO and the other emissions at various loads in different ways.[23] These tests show that increasing EGR reduces NO_x with little effect on hydrocarbons and CO up to EGR values of 20 percent, where NO_x is reduced by about 20 percent at low-load and 30 percent at the higher load. Increasing EGR above 20 percent results in an increase in hydrocarbon and CO emissions. Smoke starts to increase sharply at 40 percent EGR at light-load and at 25 percent EGR at the higher load. These results indicate that to achieve low NO emissions without increasing the other pollutants, the amount of EGR should be optimized and decreased as the engine load increases.



Source: Reference 71

FIGURE 6. EFFECT OF EXHAUST RECIRCULATION, 6-CYCL., 4-STROKE DIRECT INJECTION ENGINE - 610 TOTAL PISTON DISPLACEMENT

Recent results reported by V.W.[72] showed that EGR increases smoke, particulates, and odor, and results in poor driveability, durability and maintenance.

4.1.4 Turbocharging and Turbocharging-Intercooling

Turbocharging-intercooling tends to reduce NO emissions slightly. If combined with EGR, it will greatly reduce NO emissions, produce more power without increasing hydrocarbons, CO or smoke, and improve fuel economy. To avoid the high peak pressures caused by turbocharging, the compression ratio may be reduced but this would produce cold startability problems.[23] Parker and Walker[78] found that combining turbocharging-intercooling with retarded timing increases the power and lowers the emissions index slightly.

V.W.[72] reported that turbocharging offers definite advantages in terms of fuel economy, emissions, noise, and vehicle packaging.

4.1.5 Compression Ratio

Tests on various compression ratios in I.D.I. engines were made by Torpey et al.[74] They reported that increasing the compression ratio resulted in lower emission levels and lower noise (particularly at retarded timings), and in slightly better brake thermal efficiency and power output. Reducing the compression ratio increased noise, gave difficult starting, and increased the hydrocarbon emissions at light load. In a D.I. experimental single cylinder engine, reducing the compression ratio reduced NO, but increased smoke.[79]

4.1.6 Pilot Injection

Pilot injection is an effective way to reduce the ignition delay and high rates of pressure rise at the start of injection. This results in lower NO and noise, but in high hydrocarbon emissions.[20]

4.1.7 Fumigation

Similar to pilot injection, fumigation decreases the ignition delay, NO and noise, but produces higher hydrocarbon emissions.

4.1.8 Combustion Chamber Design

Because of the better fuel economy and lower thermal loading of the D.I. engine, some new approaches in the design of its combustion chamber have recently been reported. The first approach is to change the geometry of the bowl in the piston top. In the design by Bertodo et al.,[80] most of the clearance volume is in the form of a flat-bottomed bowl, and has a narrow throat. Bertodo et al. feel that with this design they can achieve better mixing and fast diffusion burning. Pischinger[71] found that by increasing the combustion bowl diameter, and with correct nozzle design and proper quality control in nozzle manufacture, the high hydrocarbons which are emitted during high speed idling can be reduced.

The second approach is reported by Kihara et al.[50] in which a square toroidal combustion chamber is used in medium duty engines instead of the conventional round toroidal chamber. The advantages of this design as reported by Kihara et al. are a reduction in engine noise and the ability to retard injection timing without producing the increase in smoke common to the round chamber design.

4.1.9 Electronic Controlled Fuel Injection

Electronic fuel injection is used to improve control of the injection process, to increase the thermal efficiency of the diesel engine,[81] and to eliminate secondary and irregular fuel injection.[50] This promises to be an effective tool to control diesel combustion and deserves further research and development.

4.2 AFTER TREATMENT

Most of the exhaust catalytic treatment devices have been used in diesel engines operating in mines. Marshall et al.[10] reported that four catalysts varied widely in oxidation efficiency at light to intermediate engine loads, but that all were very effective in reducing emissions of CO, HC, aldehydes, and odor intensity at heavy loads (engine operating modes resulting in high exhaust temperatures). They found that particulate loading, particulate size, and NO_x emissions were generally unaffected by any of the four catalysts. However, the catalyst increased the sulfate emissions. The exhaust treatment techniques used to reduce NO in gasoline automobile engines employ a catalyst in the presence of CO.[82,83] These techniques are not effective in diesel engines because of the presence of excess oxygen in the exhaust, even under full load conditions.

4.3 FUEL MODIFICATION

4.3.1 Fuel Additives

Nitrogen-containing fuel additives are sometimes used to increase the cetane number of the fuel or to control carbonyl-compound and CO exhaust emissions. McCreath[84] found that the additives which decrease the ignition delay also decrease the NO_x emissions, and vice versa. The decrease in the ignition delay allows less fuel to evaporate and mix with the air before the start of combustion. In the case of nitromethane, the increase in the NO_x yield appears to be chemically motivated by products of additive decomposition rather than physically altered by changes in the ignition delay. Some metallic fuel additives may reduce NO while causing the emission of harmful particulates.

4.3.2 Dual-Fuel Operation

Fuels other than the regular diesel fuel can be introduced into the engine as carbureted with the intake air, or as an emulsion with the fuel. The dual-fuel operation of the automotive diesel engine is still in the very early stages of research. The use of carbureted gas and pilot diesel fuel injection has been known for many decades.[85] It has been found to reduce the exhaust smoke. More recently, Bro and Pederson[86] reported the results of an experimental investigation of methanol, ethanol, methane and ammonia as primary fuels carbureted in a high speed D.I. diesel engine. Combustion was achieved by injecting a small amount of pilot diesel fuel. The emission of smoke was reduced, but the emission of unburned fuel (hydrocarbons and/or ammonia) was increased. The emission of NO_x increased moderately using ethanol and ammonia, while methane gave a nearly 5-fold increase near the Stoichiometric mixture, and methanol gave a somewhat lower emission.

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